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RESEARCH MEMORANDUM

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SOLID TURBINE BLADE OF SIMILAR AERODYNAMIC DESIGN

By R. H. Kemp and J. Shifman

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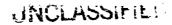
SOLID TURBINE BLADE OF SIMILAR AERODYNAMIC DESIGN

By R. H. Kemp and J. Shifman

SUMMARY

The vibrational modes of several hollow turbine blades and a solid turbine blade of similar aerodynamic design were experimentally determined. In general, several vibrational characteristics common to the hollow blades investigated were as follows:

- 1. Approximately twice as many readily excited modes were detected in the hollow blades than in the solid blade; nearly all the vibrational modes of the hollow blades could be excited if the blades were operated in a conventional turbojet engine.
- 2. An effect termed "breathing" was found to be predominant in a vibrational mode occurring at approximately 1230 cycles per second. In this mode, the two sides of the blade move alternately toward and then away from each other causing stress concentrations to be produced at the leading and trailing edges and at the rivet holes of a rivet-stiffened blade. Failure by fatigue in this mode of a hollow blade having rivets along the trailing edge produced cracks that started at the rivet holes and extended to the tip of the blade.
- 3. Nodal patterns of the vibrational modes below approximately 2600 cycles per second were similar on the two sides of the blades, whereas in the higher frequency modes the nodal patterns were dissimilar on the two sides. The hollow blades vibrated as an integral unit in the lower modes and were characterized by plate vibrations in the higher modes.
- 4. Nodal patterns of the hollow blades of same design and manufacture were similar in the lower modes (those below approximately 2600 cps) but were dissimilar in the higher modes.
- 5. No vibrational modes of the hollow blades were detected that could be designated true fundamental bending modes. The lowest frequency mode occurred at approximately 850 cycles per second.



In general, little similarity existed between the nodal patterns of the hollow blades and of the solid blade.

INTRODUCTION

The turbine blades of jet-propulsion engines fabricated in this country and in England have been almost exclusively of the solid type. The use of such blades was prompted primarily by the expediency of the rapid development of aircraft turbines. The rapidly increasing production of high-temperature-alloy turbine blades has, however, created a serious problem of raw material supply, which may make a reduction in the quantity of strategic materials used in the fabrication of turbine blades essential. The use of hollow turbine blades provides a possibility of considerable savings in strategic materials. This savings may be effected not only by use of a smaller quantity of material but by use of other less strategic material that can be adequately cooled because the blades are hollow (reference 1). In addition, hollow turbine blades permit a considerable reduction in the weight of the turbine wheel because of the smaller stresses imposed on the wheel by the lighter blades.

During the development of hollow turbine blades by several aircraft-engine manufacturers, failures were experienced that were traced to excessive vibrational stresses. A research program has therefore been undertaken at the NACA Lewis laboratory to investigate the stresses and the thermal conditions that limit the service lives of hollow blades. As part of the general program, the preliminary experimental investigation reported herein was made to determine the vibrational characteristics of several hollow turbine blades and to compare them with the vibrational characteristics determined for a solid blade of approximately the same external dimensions and airfoil contour. The effects of flutter are not considered in this report.

APPARATUS AND PROCEDURE

Description of turbine blades. - Turbine blades of the three types for which vibrational modes were determined are shown in figure 1. All blades were approximately 4 inches long and had approximately the same dimensions and airfoil contours. The trailing edges of the hollow blades were thicker than the trailing edge of the solid blade due to the minimum thickness being equal to twice the wall thickness.

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The two hollow blades of elementary design for which data are presented were chosen principally because of the ease with which they could be procured. Hollow-blade type A was plain with neither damping nor stiffening features and represented the most simple form of hollow blade of the aerodynamic design investigated having wall-thickness taper. The blade-wall thickness varied from 0.074 inch at the base to 0.024 inch at the tip. The material used in the fabrication of the blade was N-155 alloy sheet, which was formed into a cylinder and welded at the butting edges. Wall-thickness taper was then obtained by mounting the cylinder on a mandrel and machining it. The tube was filled with a hard wax and pressed between dies to the finished airfoil contour. The vibrational modes of one blade of type A were obtained.

Blades of type B were similar to type A except that two stiffening features were incorporated (fig. 1). Three rivets were installed along the trailing edge of each blade to prevent relative motion of the sides of the blade and to supply additional damping. Also, the thickness of the blade wall was increased to 0.055 inch over a portion of the blade extending from the tip to a peripheral line approximately 1/4 inch below the tip. This thickening made the tip section more rigid and, consequently, acted to reduce relative motion of the sides of the blade. The wall thickness was 0.062 inch at the base and 0.024 inch at the tip below the thickened band. Vibrational modes of four blades of this type, designated B1, B2, B3, and B4, were obtained.

Blade type C (fig. 1) was a solid Vitallium blade fabricated by precision casting. The vibrational modes of one blade of this type were obtained.

Excitation. - Resonant vibration of the turbine blades was produced by the apparatus shown in figure 2. A partly split turbine wheel with fir-tree serrations was used to hold the turbine blades for individual investigation. A turbine blade was inserted in the fir-tree serration through which the slit in the wheel passed and then the two portions of the turbine wheel were tightly clamped together by bolts. The wheel was suspended near a speaker-type exciter, which was used to produce the various modes of vibration of the blades, and was connected to the moving coil of the exciter with a metal rod. The frequency of excitation was varied by a variable-beat-frequency oscillator. The probe of a crystal-type pickup was drawn back and forth across the blade at each of the resonant conditions. The output of the crystal pickup was connected to the Y-axis of a cathode-ray oscilloscope and a small proportion of the output of the beat-frequency oscillator connected to the X-axis. The nodal patterns of each mode were accurately plotted by

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observation of the amplitudes and phase relations of the traces on the oscilloscope. When sufficient points had been plotted on the blade to determine the nodal pattern, lines were drawn to connect the points. This procedure was followed for each side of each blade.

RESULTS AND DISCUSSION

Significance of nodal patterns. - The vibrational modes of the hollow blades and of the solid blade are shown in figures 3 to 8. The exciting frequency in cycles per second is given below each nodal pattern. The solid lines in these figures represent the node lines on the concave side of the blade, and the dashed lines represent the node lines on the convex side of the blade. The node lines represent the locus of points of minimum amplitude, and by study of these lines the manner in which the specimen is vibrating may be determined. In some of the simple modes the locations of the maximum stress may be inferred from these patterns, although the node lines do not necessarily represent the locations of either the minimum or the maximum stresses.

Vibrational modes of blade A. - The nodal patterns determined for the plain hollow blade A are shown in figure 3. The lowest mode, excited at a frequency of 845 cycles per second (fig. 3(a)), has a node line running from the tip toward the base of the blade. Although this mode has the lowest frequency, which appears to be a beam vibration, it cannot be designated fundamental bending. In addition, no other detectable mode could be called fundamental bending.

The first five vibrational modes, figures 3(a) to 3(e), are representative of beam-type vibrations having similar nodal patterns on both sides of the blade. The modes observed at frequencies above 2590 cycles per second have, in general, dissimilar nodal patterns on the two sides of the blade. In these modes, the patterns become more complex at the higher frequencies and are representative of plate-type vibrations in which the blade vibrates as two plates fastened together at two sides, fixed at a third side, and free at the fourth side. Because the leading and trailing edges do not represent discontinuities, the node lines running to these edges do not stop but continue to the other side of the blade or travel along the edge to the base or the tip. The base and the tip, however, are discontinuities and node lines appearing on one side at the base or the tip do not necessarily appear on the opposite side. Although the node lines are generally discontinuous, a few are continuous. An example of a continuous node line is shown in figure 3(k) where a

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closed loop is formed; that is, the node line completely encircles the blade. Node lines that are discontinuous may have their ends at either the base or the tip and may travel around the blades several times as typified in figure 3(o).

An effect termed "breathing" occurs predominantly in the mode shown in figure 3(c) at a frequency of 1230 cycles per second. In this vibrational mode, the two sides of the blade move alternately toward and away from each other due to the bending of the blade about the node line. This action is illustrated in figure 9. Severe stress concentrations in the leading and trailing edges are produced by this action.

Vibrational modes of blades B_1 , B_2 , B_3 , and B_4 . - The nodal patterns determined for the stiffened hollow blades B1, B2, B3, and B4 are shown in figures 4, 5, 6, and 7, respectively. The lowest frequency modes of 890, 880, 850, and 880 cycles per second in figures 4 to 7, respectively, represent a variation of approximately 4.5 percent. Again, the lowest frequency mode is not a true fundamental bending mode. The first six corresponding modes of blades B_1 , B_2 , and B_3 are similar both in nodal patterns and in frequencies: the small differences observed are attributed partly to dimension and shape variations inherent in the manufacturing process and partly to slight variations in the clamping of the blades. An additional mode, in the case of blade B4, was observed at a frequency of 1490 cycles per second (fig. 7(d)). Because the amplitude to which this mode could be excited was very low, it is possible that it existed in the cases of blades B1, B2, and B3, but with an amplitude too small to detect. With the exception of the mode in figure 7(d) of blade B_A , the first six modes of blades B_1 , B_2 , and B_3 (figs. 4(a) to 4(f), 5(a) to 5(f), and 6(a) to 6(f), respectively) are similar to the modes of blade B_4 (figs. 7(a) to 7(c) and 7(e) to 7(g)). The vibrational modes occurring below 2600 cycles per second have nodal patterns on the concave side similar to those on the convex side. Above 2600 cycles per second, plate vibrations generally exist, the nodal patterns being dissimilar on the two sides and becoming more complex with increasing frequency. Also, above 2600 cycles per second there is little similarity in the nodal patterns of the four blades.

The breathing effect was found to be prominent in the B-type blades in the same mode (approximately 1230 cycles per second) as determined for the A-type blade. In the B-type blades, the alternate movement of the sides toward and away from each other also produces

a stress concentration at the rivet holes. Failure by fatigue in this mode produced cracks that started at the rivet holes and extended to the tip of the blade. Because the stiffening band at the tip of the blade neither shifts the natural frequency nor appreciably changes the nodal pattern and because the presence of the rivet holes constitutes a severe stress concentration, the advantages of the stiffening features are doubtful. Use of the stiffening band alone would possibly be more beneficial in this mode inasmuch as the points of severe stress concentration would then have greater strength.

Vibrational modes of blade C. - The nodal patterns determined for the solid blade C are shown in figure 8. This blade had the same dimensions and airfoil contour as blades A and B except for a thinner trailing edge. The lowest frequency mode occurred at 1270 cycles per second and can be considered a true fundamental bending mode. The first torsional mode occurred at 2050 cycles per second. Little similarity exists between the nodal patterns of the solid and the hollow blades, but, as might be expected, the nodal patterns on the respective sides of the solid blade were similar in all modes. Fewer vibrational modes were detected in the solid blade than in any of the hollow blades. Also, at the higher frequencies, modes of the solid blade did not appear as complex as those of the hollow blades. In this investigation, all the modes in which the blades are capable of vibrating were not determined. Modes of amplitudes too small to determine the nodal patterns accurately were detected in all blades. There probably are modes that would occur above the frequency limitation of the equipment, which was approximately 11,000 cycles per second.

Probability of excitation of blades A and B in an engine. - The excitation orders present in turbojet engines have been shown to vary over a wide range. Reference 2 indicates the presence of strong 7th, 8th, 1lth, 14th, and 48th orders in a particular engine. Inasmuch as the configuration of this engine is typical of conventional turbojet engines, nearly all the vibrational modes determined for hollow blades A and B could be excited during engine operation. Nearly all the vibrational modes determined for blades A and B must therefore be assumed to represent potential sources of fatigue failure. On the basis of the results presented, it is improbable that blades of the types represented by blades A and B could be designed with all resonant vibration responses outside the operating range of the engine. The amplitudes of those modes excited must therefore be limited to values that do not produce excessive stresses in the blades.

Vibration amplitudes may be limited by decreasing flow irregularities in the gas stream about the blade or by increasing blade damping. In general, reduction of the magnitude of the gas-stream fluctuations would require extensive changes in the engine configuration and is therefore considered impractical. The total blade damping could be increased by installing friction members that would rub against the blade walls when the blade is excited into vibration. Strengthening of the basic hollow shell may be accomplished by several methods. One means, as described in reference 1, (fig. 10) employs fins that are fabricated integrally with the blade walls. Such a design would also effect more efficient use of the cooling medium.

SUMMARY OF RESULTS

The vibrational characteristics of several hollow turbine blades were experimentally determined and compared with characteristics determined for a solid blade. The comparison of the vibrational modes of the hollow blades and of the solid blade showed little similarity. Approximately twice as many readily excited modes were detected in the hollow blades as in the solid blade in the frequency range of 0 to 11.000 cycles per second.

The fundamental bending mode of the solid blade occurred at 1270 cycles per second, whereas no vibrational modes of the hollow blades were detected that could be designated true fundamental bending modes. The lowest frequency mode of the hollow blades was approximately 850 cycles per second. A breathing effect was found to be predominant in a mode of the hollow blades occurring at approximately 1230 cycles per second. In this mode the two sides of the blade move alternately toward and away from each other causing stress concentrations at the leading and trailing edges and at any stress raisers. Failure by fatigue in the breathing mode produced cracks that started at the rivet holes of a rivet-stiffened blade and extended to the tip of the blade.

The nodal patterns on the two sides of the hollow blades were similar for modes below 2600 cycles per second but were dissimilar at the higher frequencies. The hollow blades apparently vibrated as an integral unit in the lower modes and were characterized by plate vibrations in the higher modes. The nodal patterns of hollow blades of the same design and manufacture were similar in the lower modes (below 2600 cps) but were dissimilar in the higher modes.

A large proportion of the vibrational modes of the hollow blades would probably be excited during operation in a turbojet engine of

conventional design. Limitation of the vibration amplitudes to values that would not cause excessive stresses in the blades would have to be accomplished either by reducing the magnitude of the gas-stream fluctuations, by increasing the total damping of the blades, or by strengthening the blades.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio.

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- 2. Morgan, W. C., Kemp, R. H., and Manson, S. S.: Vibration of Turbine Blades in a Turbojet Engine during Operation. NACA RM E7118, 1948.

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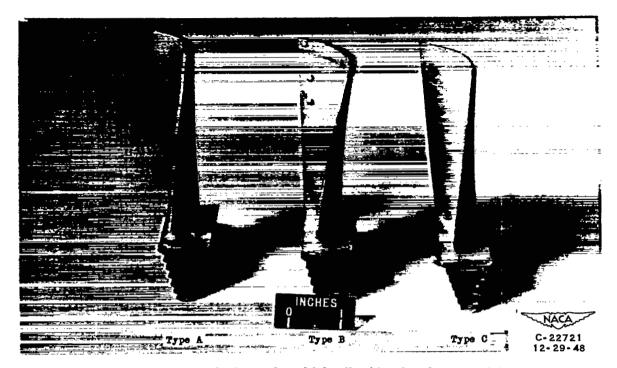


Figure 1. - Turbine-blade types for which vibrational modes were determined.

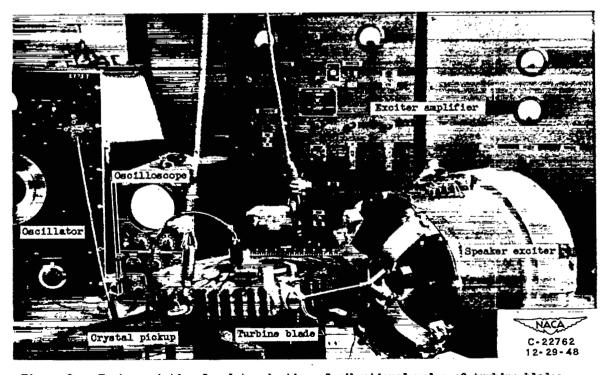


Figure 2. - Instrumentation for determination of vibrational modes of turbine blades.

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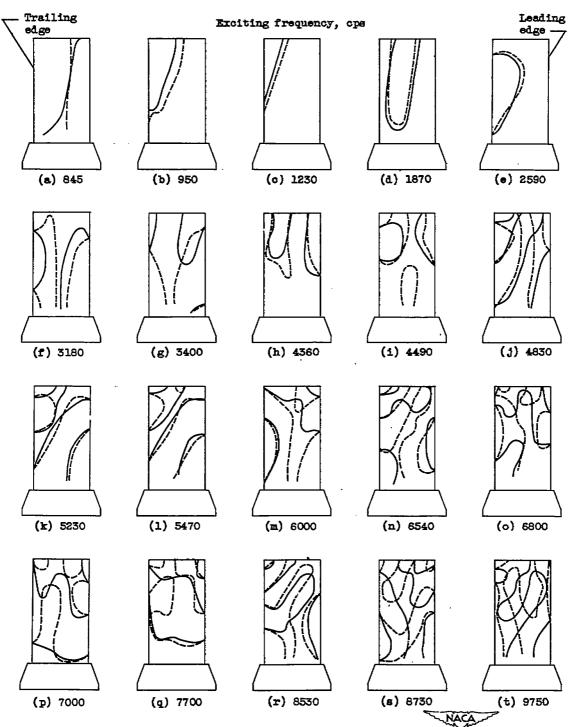


Figure 3. - Vibrational modes of plain hollow blade A. (Solid lines represent node lines on concave side of blade; dashed lines represent node lines on convex side. Exciting frequency in cycles per second is shown below each nodal pattern.)

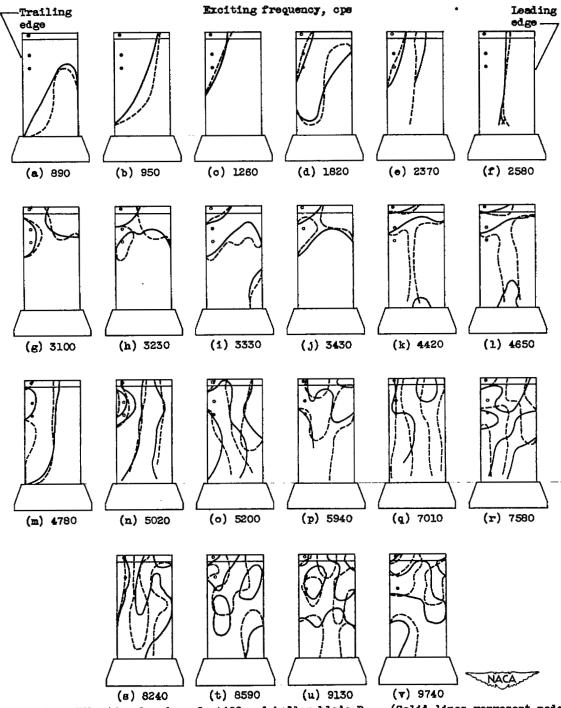


Figure 4. - Vibrational modes of stiffened hollow blade B_1 . (Solid lines represent node lines on concave side of blade; dashed lines represent node lines on convex side. Exciting frequency in cycles per second is shown below each nodal pattern.)

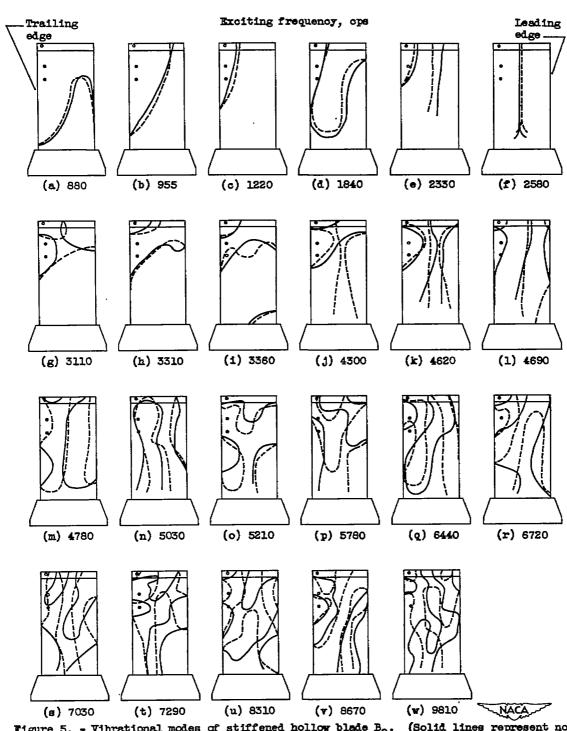


Figure 5. - Vibrational modes of stiffened hollow blade B₂. (Solid lines represent node lines on concave side of blade; dashed lines represent node lines on convex side. Exciting frequency in cycles per second is shown below each nodal pattern.)

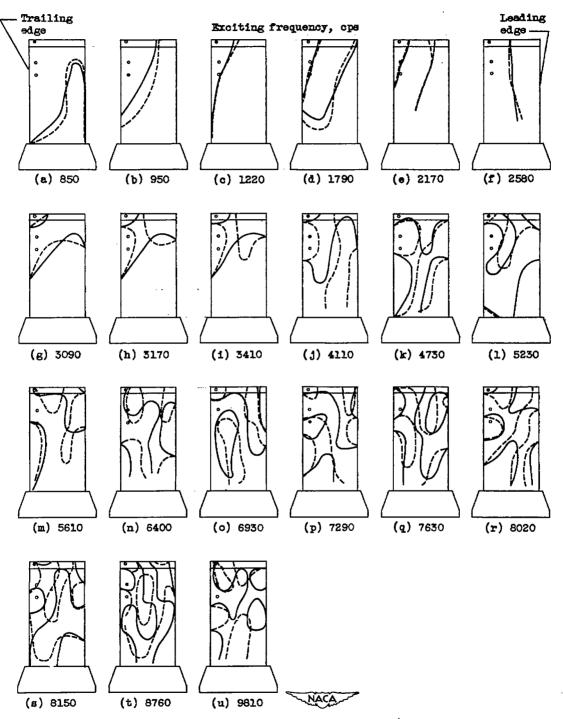


Figure 6. - Vibrational modes of stiffened hollow blade B₃. (Solid lines represent node lines on concave side of blade; dashed lines represent node lines on convex side. Exciting frequency in cycles per second is shown below each nodal pattern.)

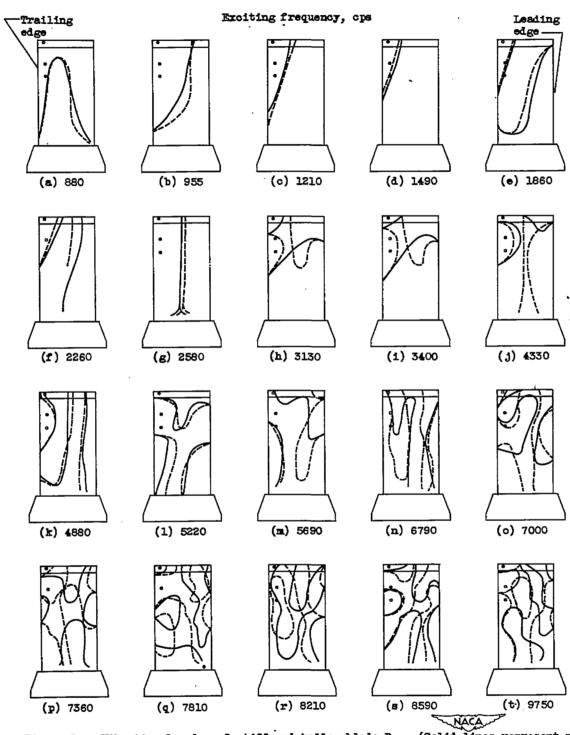


Figure 7. - Vibrational modes of stiffened hollow blade B₄. (Solid lines represent node lines on concave side of blade; dashed lines represent node lines on convex side. Exciting frequency in cycles per second is shown below each nodel pattern.)



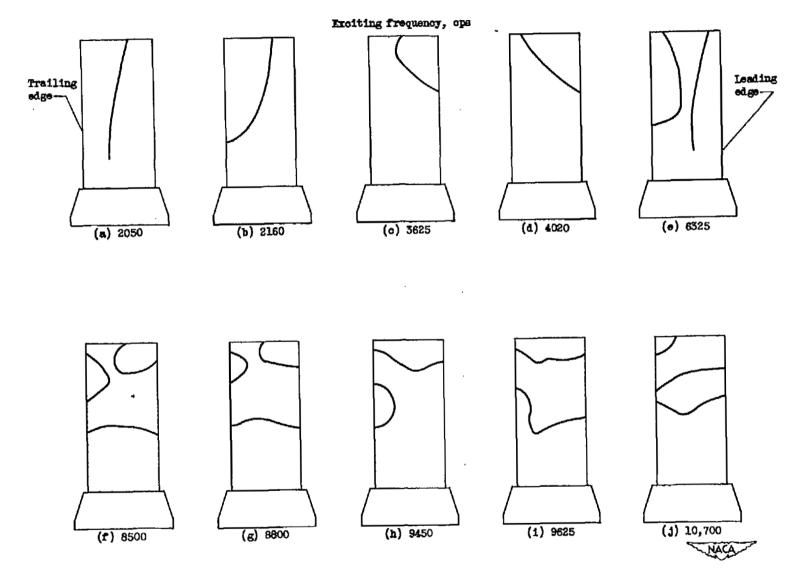


Figure 8. - Vibrational modes of solid blade C. Frequency of fundamental bending mode, 1270 cycles per second. (Exciting frequency in cycles per second is shown below each modal pattern.)

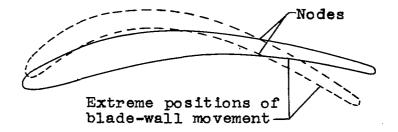


Figure 9. - Cross-sectional view showing breathing effect in basic hollow blade.

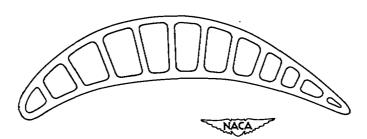


Figure 10. - Section of basic hollow turbine blade strengthened by integral fins (reference 1).

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